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(71) Applicant *(for all designated States except US)*: EMPRESA BRASILEIRA DE COMPRESSORES S/A. - EMBRACO [BR/BR]; Rua Rui Barbosa 1020, 89200-Joinville, SC (BR).

(72) Inventors; and
(75) Inventors/Applicants *(for US only)* : KRUEGER, Manfred [BR/BR]; Rua Carlos W. Boehm 864, 89200-Joinville, SC (BR). WAGNER, Nilson [BR/BR]; Rua Capinzal 58, 89200-Joinville, SC (BR).

(74) Agents: ARNAUD, Antonio, Maurício, Pedras et al.; Rua José Bonifácio 93, 8th Floor, 01003-901-São Paulo, SP (BR).

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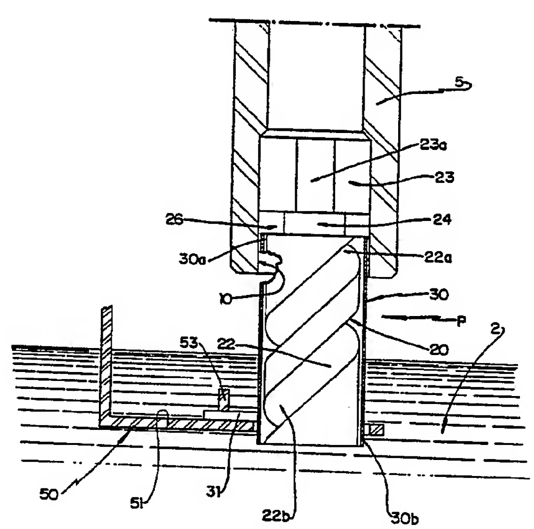
(54) Title: OIL PUMP FOR A VARIABLE SPEED HERMETIC COMPRESSOR

(57) Abstract

Oil pump for a variable speed hermetic compressor including: a hermetic shell (1) defining a lubricant oil sump (2) at its bottom and lodging, therewithin, a bearing (4) for supporting a vertical eccentric shaft (5) and a motor (6), the eccentric shaft (5) being provided with at least one channel (9) for the passage of oil, having a lower end (9b) opened to the lower end (5b) of the eccentric shaft (5) and an upper end (9a) opened to the external part of the upper median portion of the eccentric shaft (5), in the bearing region, said oil pump further comprising: at least one cylindrical extension (20), with an upper part concentrically attached to the eccentric shaft/rotor assembly, so as to rotate therewith, and provided with at least one helical peripheral groove (22), having a lower end (22b) permanently immersed in sump (2), and an upper end (22a), in fluid communication with the lower end (9b) of at least one channel (9) of the eccentric shaft (5); and a tubular sleeve (30), which is attached to the stator (7) surrounding, with a slight radial gap, at least the cylindrical portion (20) provided with the helical groove (22), said helical groove (22) being arranged in such a way as to draw the oil from the sump (2), along the internal wall of sleeve (30).



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OIL PUMP FOR A VARIABLE SPEED HERMETIC COMPRESSOR

Technical Field

The present invention refers to an oil pump for variable speed hermetic compressors of the reciprocating type, specifically those compressors provided with a vertical shaft and employed in refrigerators and freezers.

Reciprocating hermetic compressors are commonly applied in refrigerators and freezers, in which the compressor works at a constant speed of 50 or 60 Hz, according to the frequency of the local electrical network, and stops by action of a thermostat when the internal temperature of the refrigerator/freezer reaches the predetermined level.

Background Art

Advanced techniques for refrigerating systems require that the compressor supplies exactly the refrigerating capacity needed by the refrigerator, either to remove the heat infiltrated through the walls of the refrigerator, or to remove the heat introduced by hte food added or replaced inside the refrigerator. Since the capacity of the refrigerator is proportional to the flow of refrigerant mass pumped by the compressor, a variation in the refrigerating capacity involves a variation in the mass flow pumped by the compressor.

One technique to obtain such variation of the mass flow in a continuous way is to vary the speed of the motor.

There are studies indicating that variable speed compressors need a range of operation between 15 and 100 Hz, i.e., from 900 to 6000 rpm, in order to present a good refrigerating performance. This speed variation affects the mechanical operation of the

compressor, specially the operation of the oil pump, which is in charge of delivering oil to the bearings of the compressor mechanism and other parts requiring lubrication, like the connecting rod and the piston.

The centrifugal pumps are the best known oil pumping mechanisms for hermetic compressors, due to their low cost and adequate operation in rotations from 3000 to 3600 rpm, said rotations resulting from the frequency of the electrical network. Nevertheless, such mechanisms prove to be inoperative in low rotations.

Conventional oil pumps of the centrifugal type, such as that shown in figure 1 and presently in use, are not able to pump oil to the bearings when the compressor needs to work at low speeds.

The operative limitations of the centrifugal pump are related to the difference between its larger radius (R) and smaller radius (r), as seen in the equation below, which governs the behavior of the centrifugal pump:

$$\omega = [(2.g.h) / (R^2-r^2)]^{1/2}$$

where "h" is the required pumping height from the oil level up to the bearings; "g" is the gravitational constant; "R" is the larger radius of the pump; "r" is the smaller radius; and "ω" is the angular speed (rd/sec).

The aim to increase the efficiency of oil pumping in such compressors, by simply increasing the larger radius (R) of the pump is unfeasible, since said increase, although being substantially necessary to obtain the desired pumping, affects the external diameter of the compressor shaft and, consequently, all the manufacturing process of the compressor and the performance thereof, by causing mechanical

losses due to more friction. It should be understood that small diameter changes are not sufficient to achieve the desired degree of centrifugal pumping, in rotations close to or lower than 900 rpm. The conventional centrifugal pumps are widely used in hermetic compressors, as shown in the following patents: US 4.478. 559; US 4.569.639; DT 209.877 and FR 2.492.471.

Patent US 4.097.185 describes a centrifugal pump operating by stages, through which the whirl of oil at the bottom of the compressor sump can be reduced. A lower cavity in said pump allows the entrance of oil into the pump, and its smaller radius (r) is a function of the desired oil flow. As this smaller radius (r) increases, the performance of the pump, in terms of the pumping height, is reduced.

Other forms of centrifugal pumps are described in patents DT 209.936 and DT 2.502.567, which use a propeller pressed inside the shaft, serving as impelling means to the oil.

Patent FR 2.204.233, describes a conventional centrifugal pump, which is assembled to an eccentric shaft of a compressor, the pump mechanism being disposed at the lower part of the compressor body, whereas the motor is disposed at the upper part. This disposition allows the oil to be pumped in slightly lower rotations, due to a reduction in the required pumping height. The minimum rotation values, however, still remain much higher than the desired minimum values (about 900 rpm).

Another solution for oil pumping used in horizontal shaft compressors presents an extension provided at the shaft end, in the form of a tubular curved portion, with its upper end attached to the bearing housing, whereas its free end is immersed in the

lubricant oil sump of the compressor.

This tubular curved portion lodges a pump rotor defined by a coiled spring having overlapping coils, with an upper end connected to the shaft end of the compressor, in order to rotate therewith, and a conical lower end immersed in the lubricant oil. Although this construction provides helical tubes to conduct the oil, the pumping of said oil to the eccentric shaft and, consequently, to the different parts of the compressor requiring lubrication, is made by action of the centrifugal force, like in the conventional vertical shaft compressors, which are provided with an open conical free end, in order to allow this type of pumping. This solution does not show a good efficiency in low rotations, either, besides the fact that it can only be applied in horizontal shaft compressors.

Disclosure of the Invention

Thus, it is an object of the present invention to provide an oil pump for variable speed vertical shaft hermetic compressors of the reciprocating type, which need to work in a wide range of rotations, allowing an adequate lubrication, even in low rotations (about 900 rpm).

Another object of the present invention is to provide an oil pump, which can be constructed by a simple manufacturing and mounting process, without requiring any changes in the components of this type of known compressors, except the conventional oil pump replacement.

A further object of the present invention is to provide an oil pump, which does not generate any whirl of oil in the compressor sump, as it occurs in some conventional centrifugal oil pumps.

These and other objectives and advantages are

achieved through an oil pump for a variable speed hermetic compressor of the type including: a hermetic shell defining a lubricant oil sump at its bottom and lodging, therewithin, a cylinder block incorporating a bearing for supporting a vertical eccentric shaft provided with upper and lower ends; and an electric motor having a stator attached to the cylinder block and a rotor mounted to a portion of the eccentric shaft below the bearing, the eccentric shaft being provided with at least one channel for oil passage, having a lower end opened to the lower end of the eccentric shaft and an upper end opened to the external part of the upper median portion of the eccentric shaft, in the region covered by the bearing.

According to the invention, the oil pump comprises:

a pump rotor, presenting at least one cylindrical extension, with an upper part concentrically attached to the eccentric shaft/rotor assembly, so as to rotate therewith, said cylindrical extension being provided with at least one helical peripheral groove, developing upwardly in a direction opposite to the direction of rotation of said eccentric shaft, said helical peripheral groove having a lower end that is permanently immersed in the lubricant oil mass of the sump, and an upper end maintained in fluid communication with the lower end of at least one oil channel of the eccentric shaft; and

a tubular sleeve, which is attached to the stator of the electric motor and which surrounds, with a slight radial gap between said sleeve and pump rotor, at least the cylindrical portion of said rotor provided with the helical groove, said sleeve being kept in a concentric position by action of the oil which is filling up the gaps, said helical

groove being arranged so as to draw the oil from the sump, upwardly, through the helical groove and along the internal wall of the sleeve.

The subject oil pump presents an adequate pumping capacity in rotations up from 700 rpm, but it can also be used in rotations above 6000 rpm, without impairing its operation, thereby allowing its application in compressors mounted in the conventional manner, i.e., with the motor at the lower part of the body.

Brief Description of the Drawings

The invention will be described below, with reference to the attached drawings, in which:

Fig.1 is a diametral longitudinal section view of a prior art oil pump mounted inside a hermetic compressor, illustrating the measures h_1, h_2, R and r;

Figs.2a and 2b illustrate, respectively, an enlarged view of a prior art oil pump, during the pumping of oil in a normal angular speed (2a) and in a reduced speed (2b);

Figs. 3 and 3a are longitudinal section views of the interior of a hermetic compressor with an oil pump of the present invention, being respectively mounted to the rotor of the electric motor and to the tubular eccentric shaft of the compressor;

Figs 4a, 4b and 4c illustrate, respectively, an enlarged diametral longitudinal section view of the oil pump of the present invention; an upper view of the oil pump rotor; and an upper view of the supporting arm of said oil pump;

Fig.5 is a similar view to that of fig.4, except illustrating another embodiment for the oil pump of the present invention; and

Fig.6 is a similar view to that of fig.5, except

illustrating another construction for the oil pump of the present invention.

Best Mode for Carrying out the Invention

According to fig.1, a prior art variable speed hermetic compressor with a vertical eccentric shaft comprises: a hermetic shell 1, defining a lubricant oil sump 2 at its bottom and lodging therein: a cylinder block 3, incorporating a bearing 4 for supporting the vertical eccentric shaft 5, which is provided with an upper end 5a and a lower end 5b; and an electric motor 6, having a stator 7 attached to the cylinder block 3, and a rotor 8 mounted to a portion of the eccentric shaft 5 below the bearing 4 and defining an eccentric shaft/rotor assembly, the eccentric shaft 5 being provided with at least one channel 9 for oil passage, having a lower end 9b opened to the lower end 5b of the eccentric shaft 5 and an upper end 9a opened to the external part of the upper median portion of the eccentric shaft in the bearing region, said eccentric shaft 5 presenting its lower end 5b opened, in order to allow the fitting of a centrifugal oil pump 11, the lower end 11a of the latter being immersed in the oil mass provided in said sump 2.

In these compressors, as illustrated in figures 2a and 2b, the lubrication of the piston and other components is made by centrifugation, during the rotation of the eccentric shat/rotor assembly, said rotation being about 3000-3600 rpm.

The alternative construction presented in fig.2b allows the lubrication of the compressor components to be achieved in lower rotations than the lubrication achieved when the alternative construction of fig.2a is used. Nevertheless, in low rotations, usually lower than 2000 rpm, the

lubrication becomes marginal, whereas at still lower rotations, the lubrication of the components ceases to exist, since the column of oil formed by the centrifugal effect no longer reaches the upper end 9a of oil channel 9.

In these compressors, the efficiency of the oil pump is a function of the relation between its smaller diameter (radius r), immersed in sump 2, and its larger diameter (radius R). The closest these values are, the less will be the lubrication force of said oil pump, as already mentioned at the beginning of this report.

According to the present invention illustrated in figures 3-6, the eccentric shaft/rotor assembly supports, at its lower end 5b, an oil pump rotating together with said eccentric shaft/rotor assembly, due to the rotation of rotor 8.

In the preferred illustrated embodiment, the lower portion of the eccentric shaft/rotor assembly defines an axial cylindrical housing 10, the lateral wall thereof being formed by the lower portion of the eccentric shaft 5, which assumes a tubular shape, or by the lower portion of the axial central bore of rotor 8, when shaft 5 is a massive piece or retracted in relation to the lower face of rotor 8.

Said oil pump P comprises a pump rotor 20, of cylindrical shape, the surface thereof being provided with helical grooves 22 and a tubular sleeve 30. Said pump rotor 20 further presents a fastening upper portion 23, such as a retaining cylinder head, which is spaced from the upper face of said pump rotor 20, by a spacing element 24, usually in the form of a central neck of smaller diameter.

The cylinder head 23 restrains said pump rotor 20

from relative movements in relation to the eccentric shaft/rotor assembly, due to the pressure exerted by the lateral walls of said head 23 against the internal lateral wall of the axial cylindrical housing 10.

In another possible embodiment, pump rotor 20, which is defined by a unique body, with at least the lower portion of its superficial extension being provided with a helical groove, is attached to the eccentric shaft/rotor assembly by other known fixing means, such as nuts ands bolts, or other known element of the prior art that can exert pressure against the lateral wall of the axial housing.

The pump rotor 20 presents one or more helical peripheral grooves 22, as a function of the oil flow required to lubricate the compressor. The determination of this flow also considers geometrical parameters like pitch, width and depth of the grooves.

For a better perfomance, some of these parameters should present dimensional constructive characteristics that are able to maintain a proportional relationship to each other. In the subject invention, an improved performance with a drag-type lubrication is achieved when the grooves of pump rotor 20 present a width/depth ratio between 4 and 6.

Helical grooves 22 develop upwardly, in the direction of rotation opposite to that of rotor 8, throughout the length of pump rotor 20, from a lower end 22b of said grooves 22 that is permanently immersed in the oil mass of said sump 2, up to an upper end 22a.

In a constructive variant of the present invention, as illustrated in fig.5, pump rotor 20 presents the

beginning of said grooves 22 at a superficial portion 20b, adjacent to its free end immersed in oil. In this construction, the oil is initially pumped by centrifugation, up to said superficial portion 20b, where grooves 22 begin. Pumping occurs when the oil enters a central through-bore 25 and a transverse through-bore 27, the former extending from the lower face of pump rotor 20, up to the superficial region 20b of said rotor, where each respective groove 22 begins. The central through-bore 25 may also present a divergent upward development, in relation to the lower face of pump rotor 20. In either case, the upper end portion thereof opens to a plurality of secundary channels, leading to corresponding superficial openings on the pump rotor 20, wherefrom they proceed as helical peripheral grooves.

The provision of neck 24 creates an annular region 26, defining a pressure equalizing chamber of the oil which is being pumped to the cylinder head 23 and eccentric shaft 5, the dimensioning of said chamber 26 being such as to prevent the oil flow from suffering therein any restrictions towards the cylinder head 23, which would increase the pressure in the region of neck 24, causing leakages of the lubricant oil on the internal walls of rotor 8 or on the eccentric shaft 5 and, consequently, on the oil return to sump 2.

Thus, the diameter of said neck 24 should be smaller than the internal diameter of one region of the upper face of pump rotor 20, where the helical grooves 22 terminate, presenting a circular contour that is internally tangent to the upper end of said helical grooves 22.

In another embodiment, pump rotor 20 may communicate

with the cylinder head 23, through a plurality of axial projections of reduced diameter, which are distributed on the upper face of pump rotor 20 and limited by said circular contour tangent to the upper end of said helical grooves.

The passage of lubricant oil through cylinder head 23 takes place by means of communicating elements, such as a unique groove or a plurality of superficial longitudinal grooves 23a provided on said cylinder head 23, each groove 23a having a transversal width proportional to the width of a corresponding helical groove 22 provided in pump rotor 20.

In a possible embodiment (not illustrated), cylinder head 23 presents said communicating elements 23a between the equalizing chamber 26 and the eccentric shaft 5, longitudinally crossing the inside of its body. In this case, said elements take the form of a divergent bundle of peripheral grooves leading to said eccentric shaft 5. Said grooves 23a are disposed close to the longitudinal surface of cylinder head 23, so as to minimize the effect of centrifugal force against the oil mass that is going to reach the lower end of each groove 23a opened to the equalizing chamber 26.

In a second non-illustrated embodiment, cylinder head 23 is simultaneously provided with internal channels and longitudinal superficial grooves for the conduction of lubricant fluid to the eccentric shaft 5.

As cylinder head 23 does not have relative movements in relation to the eccentric shaft 5 and rotor 8, the oil conduction is accomplished through centrifugal force and not by drag, making unnecessary the provision of helical superficial

grooves on cylinder head 23.

In another constructive variant shown in figure 6, the communicating element between the upper end of pump rotor 20 and eccentric shaft 5 consists of an extension of said helical grooves 22 on the region of cylinder head 23. Such a conception facilitates the manufacturing process of the communicating elements situated on the cylinder head 23, due to the fact that these elements are an extension of said helical grooves 22. The differences between the diameter of cylinder head 23 and the diameter of pump rotor 20 guarantee the absence of restrictions to the oil flow when passing through said cylinder head 23.

When the eccentric shaft 5 is tubular, the oil emerging from cylinder head 23 is forced to flow upwardly on the internal walls of said eccentric shaft 5, till it reaches an oil outlet channel 9a, which is provided at the upper medium part of the body of said eccentric shaft and which sends the lubricant fluid to bearing 4 and other parts of the compressor to be lubricated during operation.

When said eccentric shaft 5 is a massive piece, the oil, after reaching the region between said eccentric shaft 5 and cylinder head 23, is lead to the upper end 9a of the oil channel 9 which, in this case, is upwardly divergent and eccentric in relation to the geometric axis of the eccentric shaft/rotor assembly, in order not to force the oil emerging from pump rotor 20 to flow against the action of centrifugal force. The upwardly divergent disposition of said channel functions like the internal wall of a conventional tubular eccentric shaft used in vertical shaft compressors. In this case, there is formed a peripheral annular film of

lubricant oil in an axial spacing provided between pump rotor 20 and eccentric shaft 5, due to the existence of a plurality of helical grooves 22 arriving at this region, as well as to the cross section of said grooves 22 and to the only oil channel provided in the eccentric shaft. This annular film is formed even in the absence of cylinder head 23. Said film need not exist when pump rotor 20 and eccentric shaft 5 are provided with a helical groove 22 and a channel 9, respectively.

When mounted to the eccentric shaft 5, pump rotor 20 penetrates partially within the axial housing 10, together with a length portion of sleeve 30 involving, with a slight radial gap, the whole superficial grooved extension of said pump rotor 20, in order to cause the drag of oil to said eccentric shaft 5 and to prevent said lubricant fluid, which has been conducted by pump rotor 20, from returning to sump 2, by flowing down the internal wall of said rotor 8 or eccentric shaft 5, thereby causing pumping power loss.

Eventually, said sleeve 30 may extend beyond the upper end face of pump rotor 20, in order to increase the oil retention effect in relation to the internal wall of said axial housing 10, said wall being defined by the tubular eccentric shaft 5 or rotor 8 itself.

Said sleeve 30 comprises a single-piece cylindrical body, having a diameter which is slightly smaller than the internal diameter of axial housing 10 of the eccentric shaft/rotor assembly, but slightly larger than that of said pump rotor 20, so as to be maintained separated from the surfaces of said elements, providing a small radial gap therebetween.

This spacing is necessary to avoid the rotation of said sleeve 30 together with rotor 8, or eccentric shaft 5, which would eliminate the oil dragging operation.

The sleeve 30, from a region adjacent to its opposite end 30b, presents a projection 31 which avoids the longitudinal and rotational motions of said sleeve 30 in relation to the eccentric shaft 5. The upper end 30a of said sleeve 30 is surrounded by the lower edge of the eccentric shaft 5, when the latter is a tubular shaft, or by rotor 8, in the case of a massive eccentric shaft, or further by a tubular eccentric shaft, which is not completely extended along the internal housing of rotor 8.

The longitudinal dimensioning of pump rotor 20 and sleeve 30 is made in such a way that, even during the operation of the compressor, when the oil level in sump 2 becomes lower, the lower end 30b of sleeve 30, and mainly the lower portion of pump rotor 20 are maintained immersed in the oil mass, so as not to affect the lubrication of the compressor components during operation.

This condition of immersion is achieved by dimensioning the pump rotor 20 in such a way that, to an inoperative compressor, most extension of the first coil of each helical groove 22 is found immersed in the oil mass of sump 2.

In another possible embodiment, sleeve 30 has its upper end edge leveled with the lower end of the eccentric shaft 5/rotor 8 assembly, without entering into the axial housing 10. In this case, pump rotor 20 presents its upper end incorporating at least one axial projection which is hollow and fittable in a respective oil channel 9 of a massive eccentric shaft 5, in order to communicate each helical groove

22 of pump rotor 20 with a respective oil channel 9. The lower limit of said sleeve 30 need not necessarily follow the lower end of pump rotor 20; it may extend beyond said lower end, because its function in this region of immersion is to avoid the whirling of said lubricating fluid during the operation of the compressor.

During oil pumping operation, pump rotor 20 conveys the oil through helical grooves 22, from sump 2 to the eccentric shaft 5, by drag, due to the scrape caused when the edge of each coil of helical groove 22 moves along the internal wall of sleeve 30.

In a constructive form of the present invention, sleeve 30 is supported, at its bottom, by an "L" shaped arm 50, which is fastened by proper means, such as a bolt, to the stator 7 of the electric motor 6, directly or through the cylinder block 3.

Said arm 50 comprises a straight stem portion 51 and an annular portion 52 adjacent to the lower end 30b of sleeve 30. This annular portion 52 is easily crossed by the lower end 30b of sleeve 30 and by the corresponding lower end of pump rotor 20, thus allowing the axial stabilization of sleeve 30 adjacent to the interior of sump 2.

The arm 50 further presents a tooth 53 provided at the straight stem portion 51, which is adjacent to the annular portion 52 and disposed so as to act as a stop against a radial projection 31 of sleeve 30, this latter being radially projected from a portion of the external surface of said sleeve 30, adjacent to its lower end 30b, in order to prevent said sleeve 30 from rotating together with pump rotor 20, impelled by the oil mass conveyed by pump rotor 20.

The fastening between sleeve 30 and arm 50 is achieved by the contact between the radial

projection 31 and tooth 53, occurring just after said pump rotor 20 starts to rotate, when the still unfixed sleeve 30 rotates jointly with the rotation of pump rotor 20, taken by the mass of oil conveyed by helical grooves 22.

Other constructive forms are possible for the sleeve/arm assembly, such as a construction in a single piece, or by using a supporting arm provided with an angular tooth having a perfect secure fitting into a corresponding angular tooth provided in sleeve 30, in order to restrain both pieces from relative movements, which would dispense the existence of the annular region of said arm 50. In this case, the mounting of the sleeve/arm assembly should take place before its introduction into the compressor.

CLAIMS

1. Oil pump for a variable speed hermetic compressor of the type including: a hermetic shell (1) defining a lubricant oil sump (2) at its bottom and lodging, therewithin, a cylinder block (3) incorporating a bearing (4) for supporting a vertical eccentric shaft (5) provided with upper (5a) and lower (5b) ends; and an electric motor (6) having a stator (7) attached to the cylinder block (3) and a rotor (8) mounted to a portion of the eccentric shaft (5) below the bearing (4), the eccentric shaft (5) being provided with at least one channel (9) for the passage of oil, having a lower end (9b) opened to the lower end (5b) of the eccentric shaft (5) and an upper end (9a) opened to the external part of the upper median portion of the eccentric shaft (5), in the region of bearing (4), characterized in that it further comprises:

at least one cylindrical extension (20), with an upper part concentrically attached to the eccentric shaft/rotor assembly, so as to rotate therewith as a pump rotor, said cylindrical extension (20) being provided with at least one helical peripheral groove (22), developing upwardly in the direction opposite to the direction of rotation of said eccentric shaft (5), said helical peripheral groove (22) having a lower end (22b) that is permanently immersed in the lubricant oil mass of sump (2), and an upper end (22a) maintained in fluid communication with the lower end (9b) of at least one said channel (9) of the eccentric shaft (5);

and a tubular sleeve (30), which is attached to the stator (7) and which surrounds, with a slight radial gap, at least the cylindrical portion (20) provided with the helical groove (22), said sleeve (30) being

kept in a concentric position in relation to said cylindrical portion (20), by action of the oil which is present in said radial gap, said helical groove being arranged in such a way as to draw the oil from the sump (2), upwardly, through the helical groove (22) and along the internal wall of the sleeve (30).

2. Oil pump, as in claim 1, in which said compressor further includes an axial cylindrical housing (10), whose lateral wall is defined by one of the lower portions (5b) of the eccentric shaft (5) and rotor (8), <u>characterized</u> in that the fixation of the pump rotor (20) to the eccentric shaft/rotor assembly is effected inside the axial housing (10).

3. Oil pump, as in claim 2, <u>characterized</u> in that the cylindrical sleeve (30) presents an upper portion (30a), which penetrates into the inside of said axial cylindrical housing (10), maintaining a slight radial gap relative to the lateral wall of said axial housing.

4. Oil pump, as in claim 3, <u>characterized</u> in that an helically grooved portion of said pump rotor (20) penetrates into said axial cylindrical housing (10).

5. Oil pump, as in claim 2, <u>characterized</u> in that said pump rotor (20) includes an upper fastening portion (23), which is peripherally fixed to the lateral wall of said axial cylindrical housing (10).

6. Oil pump, as in claim 5, <u>characterized</u> in that said upper fastening portion (23) is in the form of a retaining cylinder head.

7. Oil pump, as in claim 5, <u>characterized</u> in that said pump rotor (20) incorporates, at its upper part, said fastening portion (23) and presents an upper end (20a) in the axial housing (10).

8. Oil pump, as in claim 7, <u>characterized</u> in that said fastening portion (23) is incorporated to said

pump rotor(20) by a spacer defining an axial spacing between said fastening portion (23) and pump rotor (20).

9. Oil pump, as in claim 8, characterized in that said axial spacing defines an oil pressure equalizing chamber.

10. Oil pump, as in claim 9, characterized in that the upper end (22a) of said helical groove (22) terminates in a corresponding upper face of said pump rotor (20).

11. Oil pump, as in claim 10, characterized in that said fastening portion (23) is incorporated to the pump rotor (20) by means of a central neck (24) having a smaller diameter than the diameter of a maximum circular contour internally tangent to said helical groove (22).

12. Oil pump, as in claim 11, characterized in that said fastening portion (23) is cylindrical, provided with through peripheral longitudinal grooves (23a), which communicate the fluid, emerging from pump rotor (20), with the lower end (9b) of said channel (9) of the eccentric shaft (5).

13. Oil pump, as in claim 12, characterized in that said through grooves (23a) are longitudinal superficial channels.

14. Oil pump, as in claim 1, characterized in that said cylindrical sleeve (30) is attached to the stator (7) by means of an arm (50).

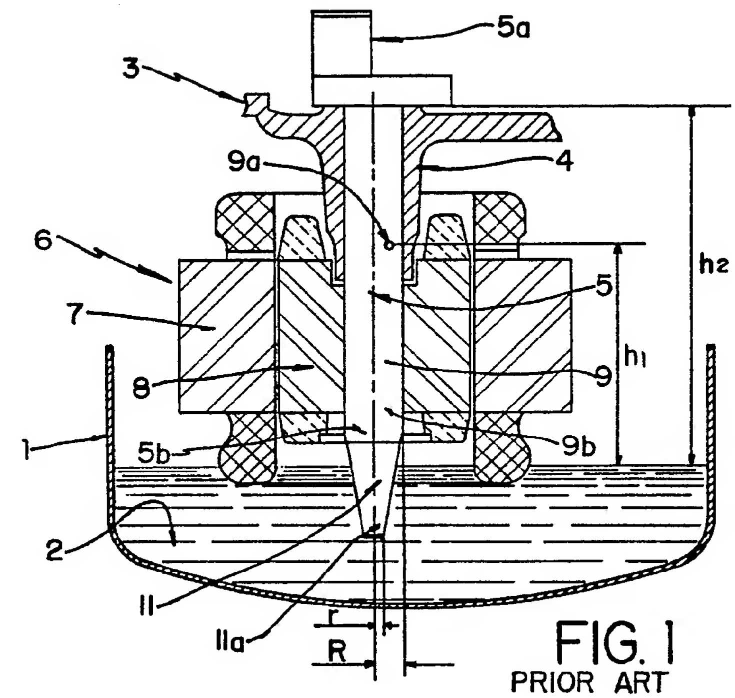


FIG. 1
PRIOR ART

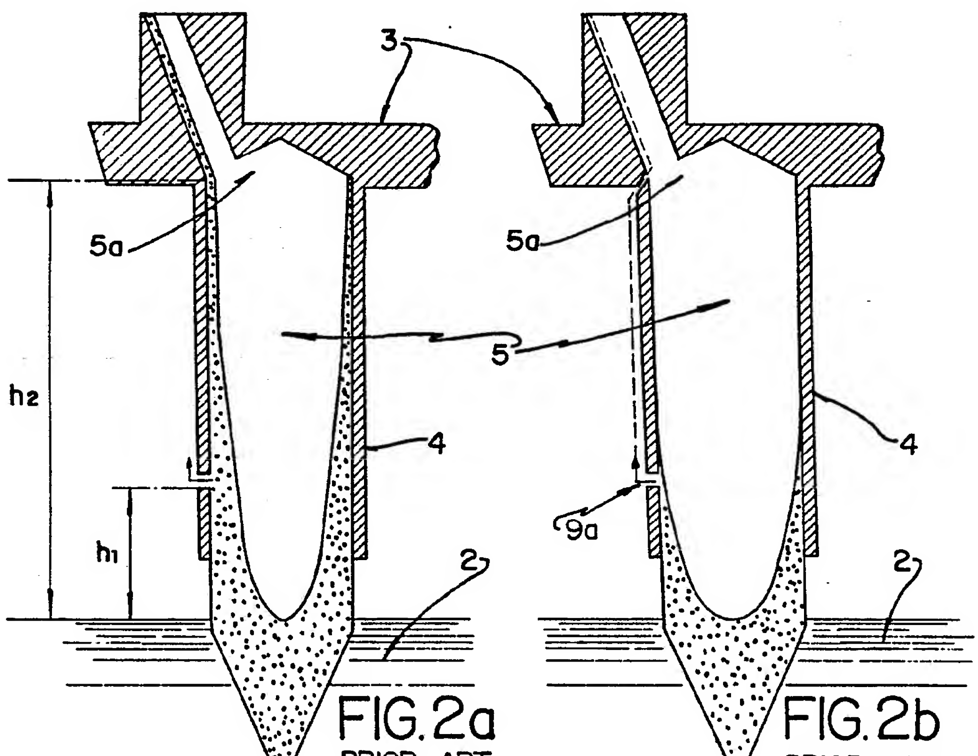


FIG. 2a
PRIOR ART

FIG. 2b
PRIOR ART

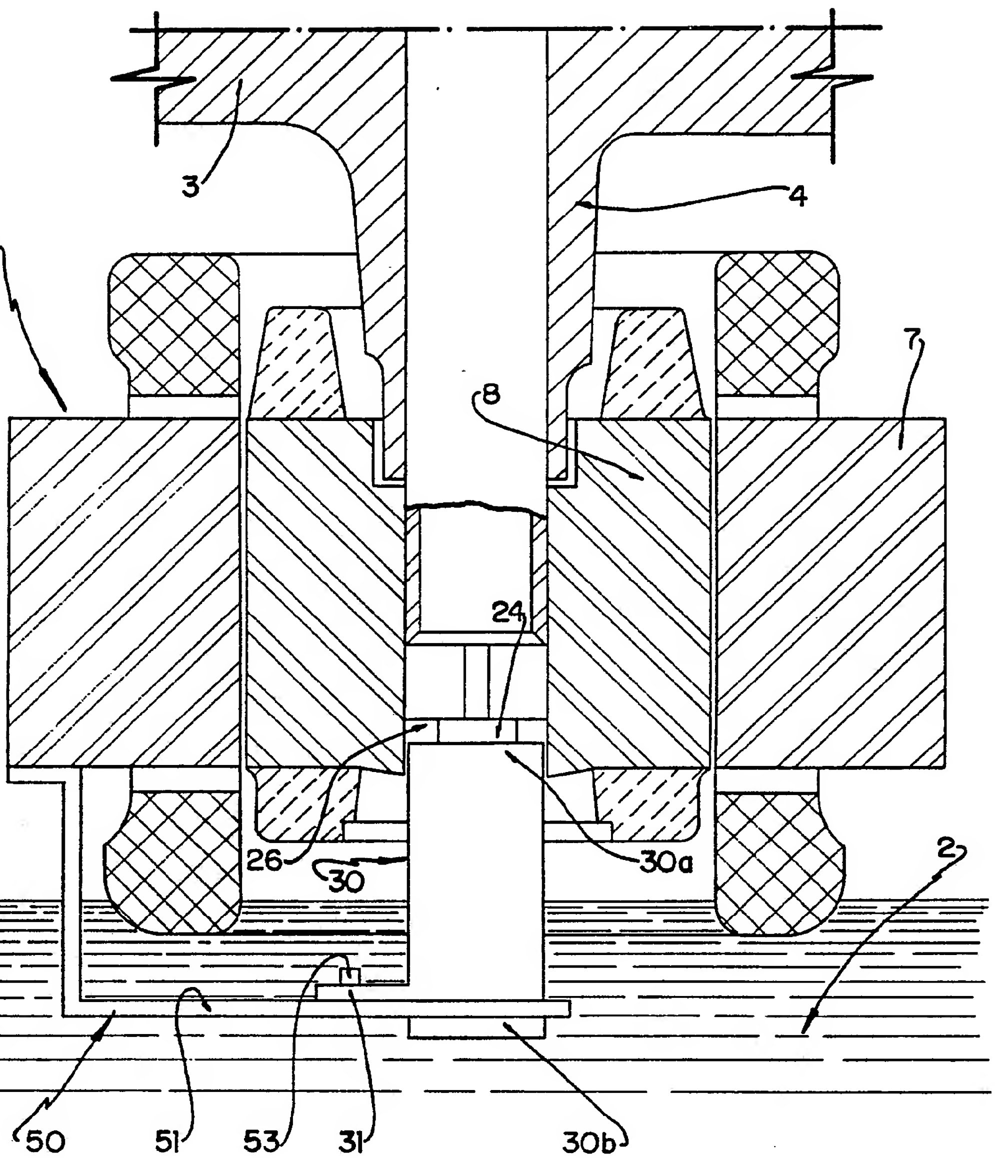


FIG. 3

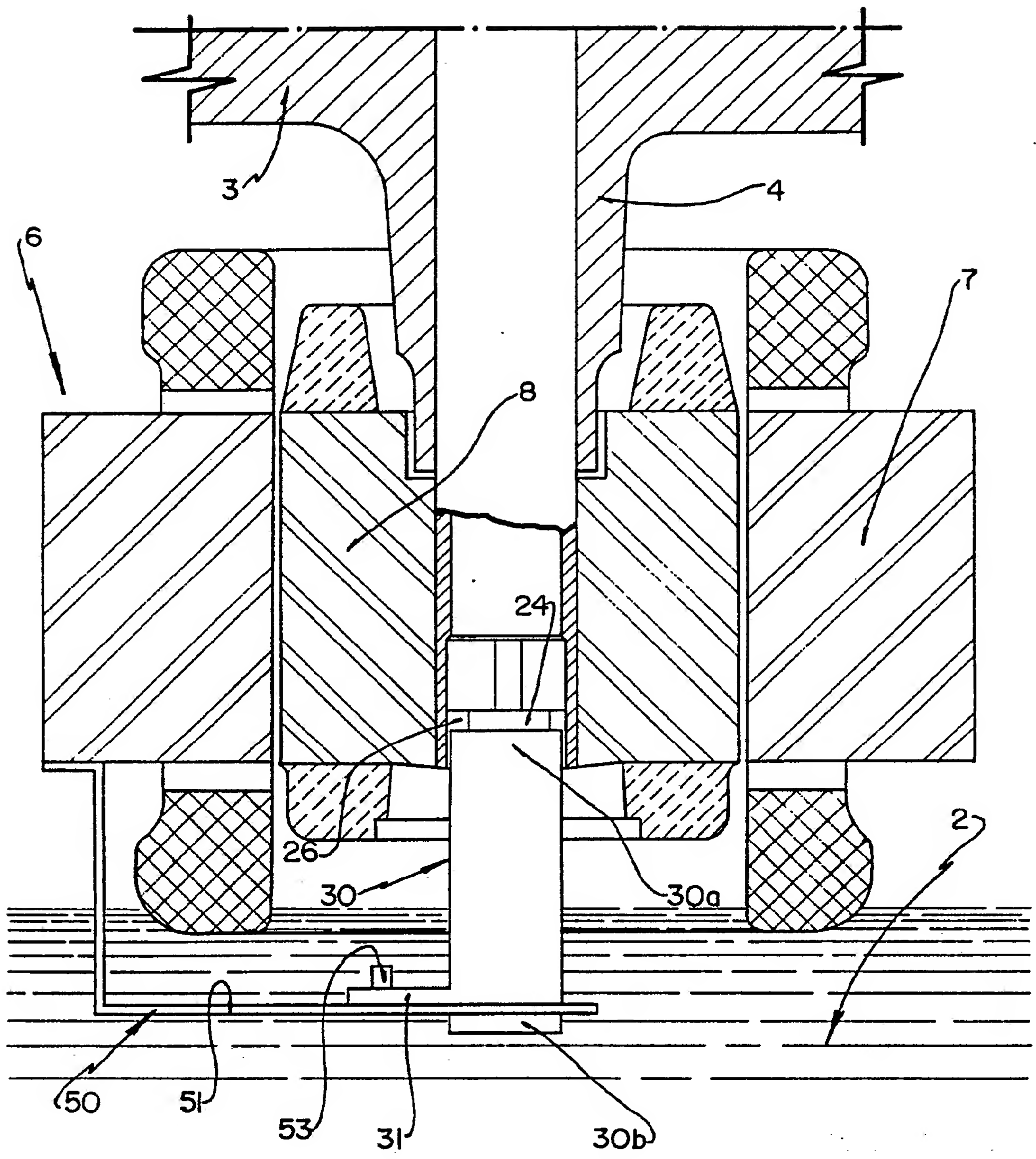


FIG.3a

FIG.4a

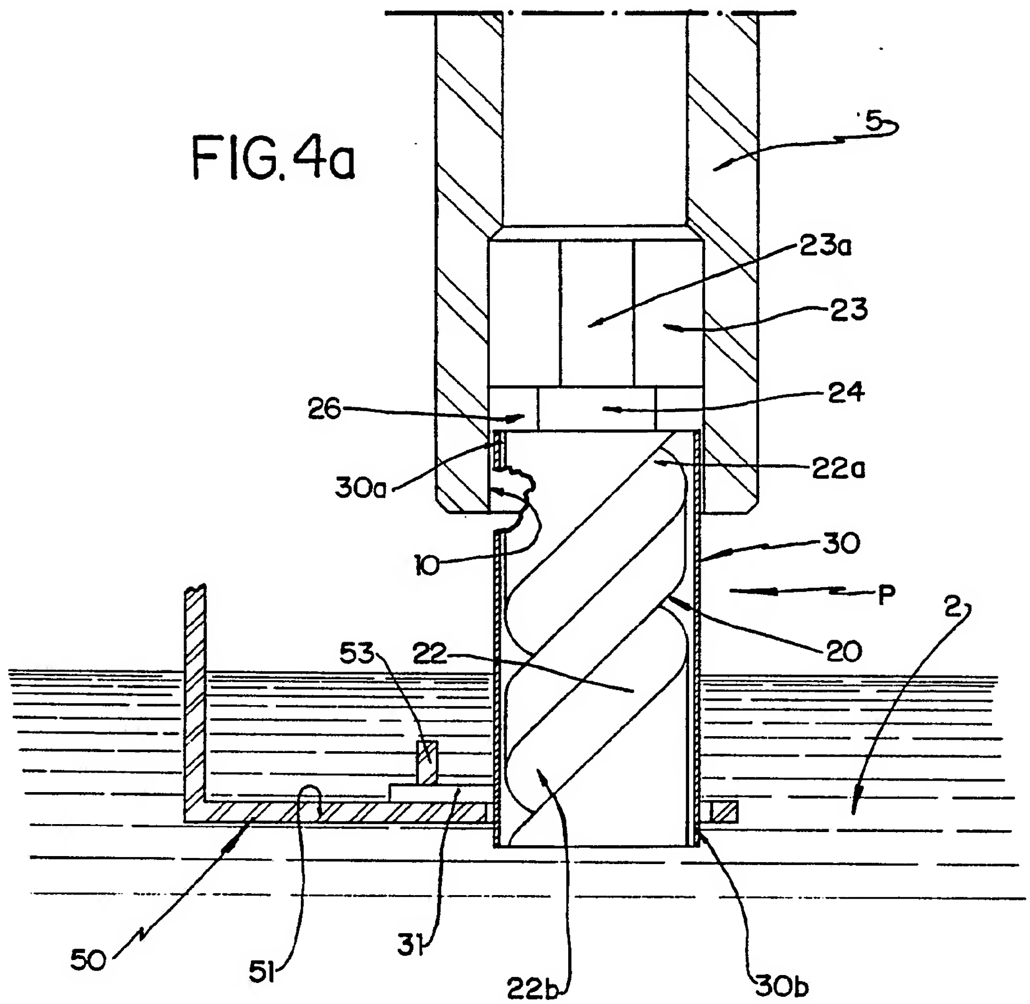
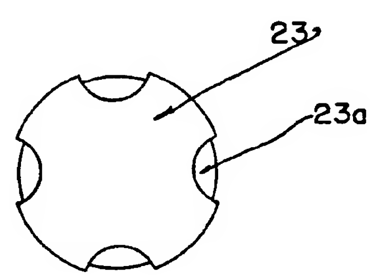


FIG.4b



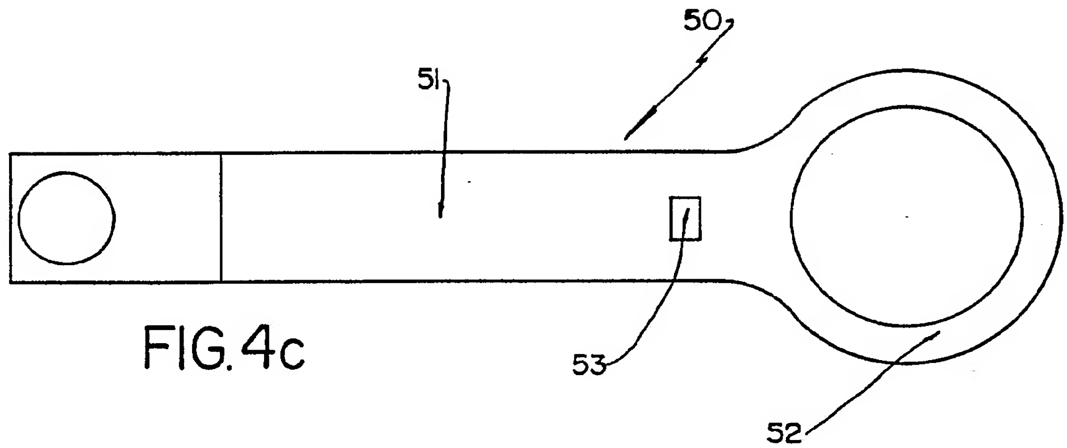
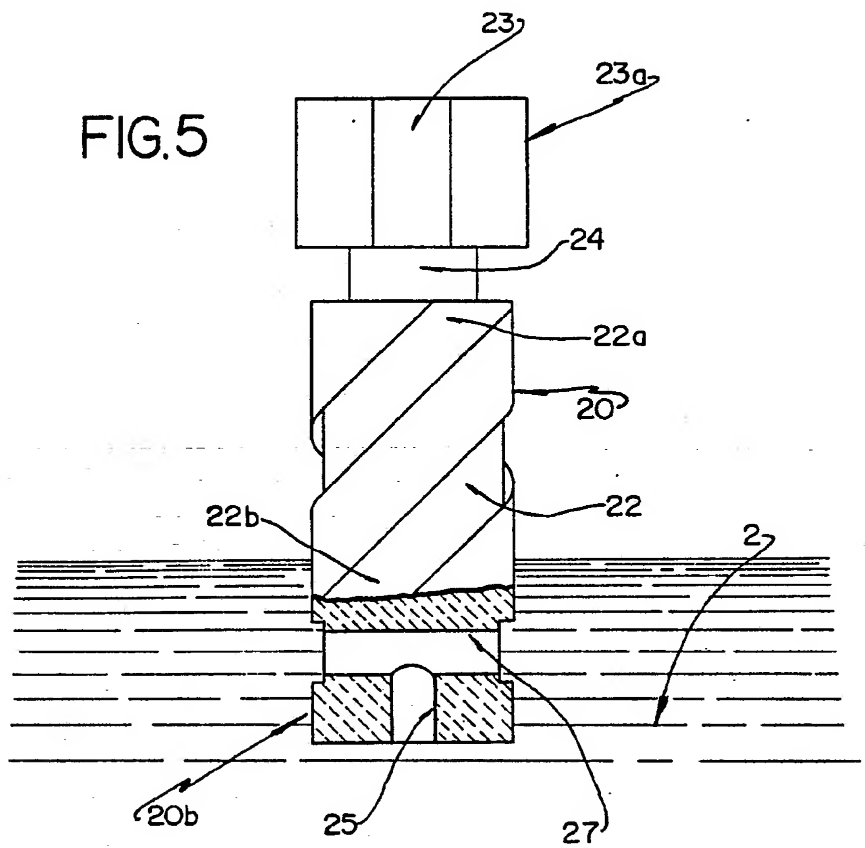


FIG. 4c

FIG. 5



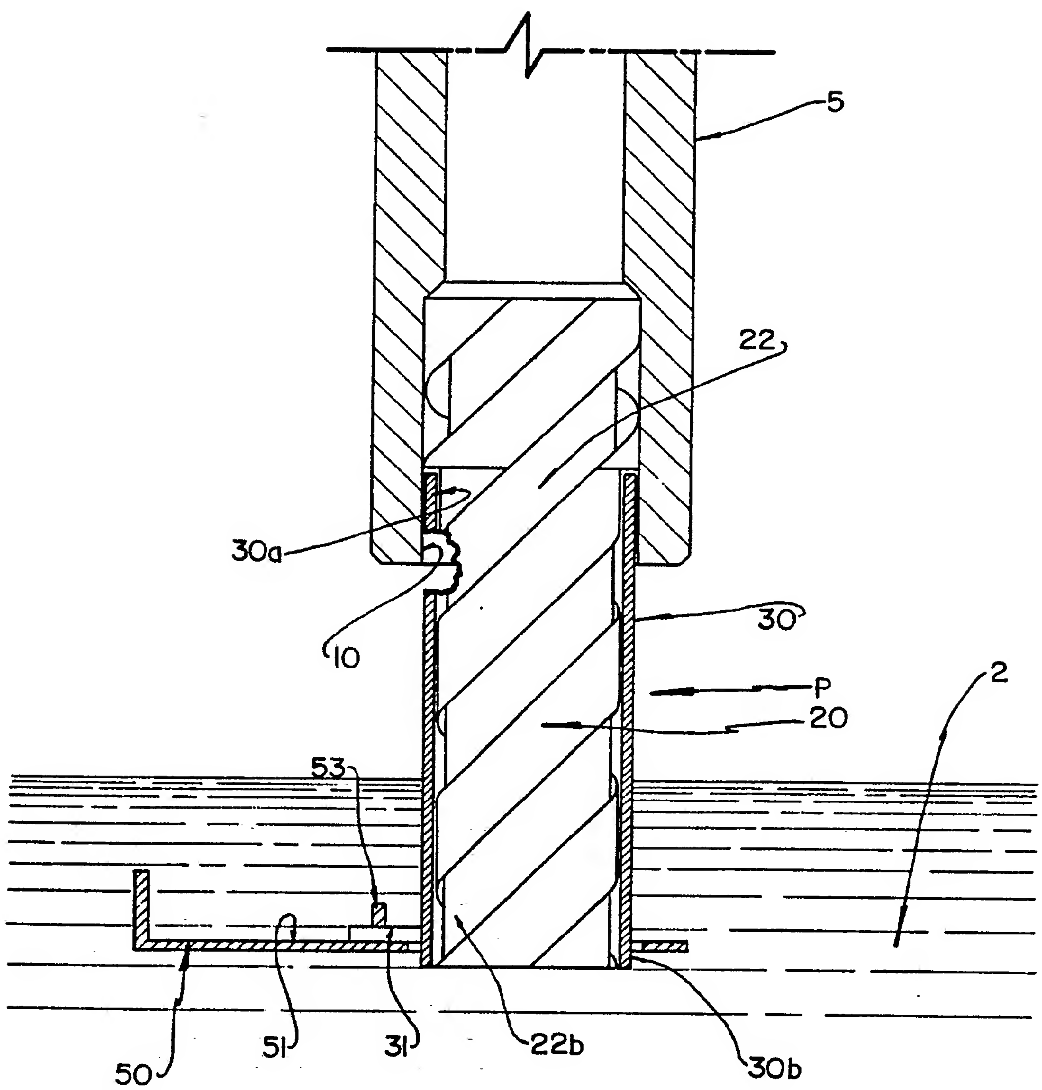


FIG.6

INTERNATIONAL SEARCH REPORT

International Application No PCT/BR 93/00016

I. CLASSIFICATION OF SUBJECT MATTER (if several classification symbols apply, indicate all)[6]

According to International Patent Classification (IPC) or to both National Classification and IPC

Int.Cl. 5 F04B39/02; F16N7/36

II. FIELDS SEARCHED

Minimum Documentation Searched[7]

Classification System	Classification Symbols
Int.Cl. 5	F04B ; F04C ; F16N

Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched[8]

III. DOCUMENTS CONSIDERED TO BE RELEVANT[9]

Category °	Citation of Document, [11] with indication, where appropriate, of the relevant passages [12]	Relevant to Claim No.[13]
X	US,A,3 182 901 (SOLOMON) 11 May 1965 see column 2, line 10 - line 65; figures 1-4 ---	1,14
X	PATENT ABSTRACTS OF JAPAN no. 9098 (M-375)27 April 1985 & JP,A,59 221 483 (TOSHIBA) 13 December 1984 see abstract ---	1,14
A	US,A,1 416 640 (HOLTE) 16 May 1922 see page 1, line 45 - line 68; figure 1 ---	1
A	US,A,2 583 583 (MANGAN) 29 January 1952 see column 1, line 42 - column 2, line 34; figures 1,3 ---	1

-/--

° Special categories of cited documents : [10]

"A" document defining the general state of the art which is not considered to be of particular relevance

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"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

IV. CERTIFICATION

Date of the Actual Completion of the International Search	Date of Mailing of this International Search Report
14 JULY 1993	27.07.93
International Searching Authority	**Signature of Authorized Officer**
EUROPEAN PATENT OFFICE	BERTRAND G.

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III. DOCUMENTS CONSIDERED TO BE RELEVANT (CONTINUED FROM THE SECOND SHEET)

Category °	Citation of Document, with indication, where appropriate, of the relevant passages	Relevant to Claim No.
A	GB,A,846 264 (ENGLISH ELECTRIC COMPANY) 31 August 1960 see page 1, line 42 - page 2, line 38; figures 1,2	1
A	GB,A,768 058 (TEVES) 13 February 1957 see page 1, line 46 - line 71; figure 1	1,2
A	US,A,4 569 639 (HANNIBAL) 11 February 1986 cited in the application see the whole document	

ANNEX TO THE INTERNATIONAL SEARCH REPORT ON INTERNATIONAL PATENT APPLICATION NO.

BR 9300016
SA 72854

This annex lists the patent family members relating to the patent documents cited in the above-mentioned international search report.
The members are as contained in the European Patent Office EDP file on
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Patent document cited in search report	Publication date	Patent family member(s)		Publication date
US-A-3182901		None		
US-A-1416640		None		
US-A-2583583		None		
GB-A-846264		None		
GB-A-768058		None		
US-A-4569639	11-02-86	AU-B-	548855	02-01-86
		AU-A-	1605783	21-11-83
		EP-A,B	0108795	23-05-84
		SU-A-	1521286	07-11-89
		WO-A-	8303878	10-11-83